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TRANSLATION

NEWS ON THE HIGHER INSTITUTIONS OF LEARNING
MACHINE CONSTRUCTION (SELECTED ARTICLES)

FOREIGN TECHNOLOGY DIVISION

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AIR FORCE SYSTEMS COMMAND

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THE COOLING OF THE TURBINES OF HIGH-TEMPERATURE GAS-TURBINE ENGINES

K. M. Popov, Candidate of Tech. Sciences

(Bauman Higher Technical School, Moscow)

Different systems of cooling the blades of gas turbines are considered in the article. It is recommended that air cooling be used at gas temperature no higher than 1400°K.

In order to increase the efficiency of liquid cooling it is advisable to use regeneration of the cooling heat.

The gas-temperature level attained in stationary and transportable gas-turbine assemblies does not exceed 1000°K. Further increase in this level is limited mainly by the strength of the basic units of the turbines, namely, the turbine wheel and the nozzle apparatus. The ultimate strength of the materials used for gas-turbine blades decreases sharply with an increase in temperature, so that the creation of turbines without cooling becomes impossible. The improvement of the heat-resistant properties of materials is taking place slowly, thus retarding to a certain extent the development of new gas-turbine designs. Therefore a great deal of attention is being given in many

countries at the present time to the problems involved in cooling turbines.

The cooling systems already known can be divided into two groups, namely, open systems and closed systems. In open cooling systems the coolant used is generally air, which is then ejected into the gas stream. In closed systems the coolant is most often a liquid, which continuously circulates in a closed circuit between a heat exchanger and the element being cooled. It is not impossible, however, to use a liquid in open cooling systems and air in closed systems.

The possible cooling systems of gas turbines are shown schematically in Fig. 1. In system a the turbine is cooled by air taken from the engine compressor. After being cooled the air is mixed with the gases in the turbine. In systems b and c the coolant circulates in closed circuits; the heat removed during the cooling of the turbine is dissipated in system b in an external radiator and is dissipated in system c in a heat-exchanger located between the compressor and the combustion chamber. Hereafter, system a will be called an open system of air cooling, system b will be called a system with external dissipation of the cooling heat, and system c will be called a regenerative cooling system. In order to calculate the quantity of heat removed during cooling of the turbine and to determine the temperature of the parts being cooled, it is necessary to know the heat-transfer coefficients in turbine cascades is made difficult by the complexity of the processes occurring during cascade flow. For example, it is difficult to determine with sufficient accuracy the point of transition of laminar flow in the boundary layer into turbulent cascade flow. Therefore at the present time experimental coefficients obtained by blowing through flat cascades are the ones mainly used for calculating heat

transfer in the cascades.

The use of these data for calculating heat transfer in nozzle cascades does not introduce great errors in the determination of the quantity of heat removed during the cooling of these cascades.

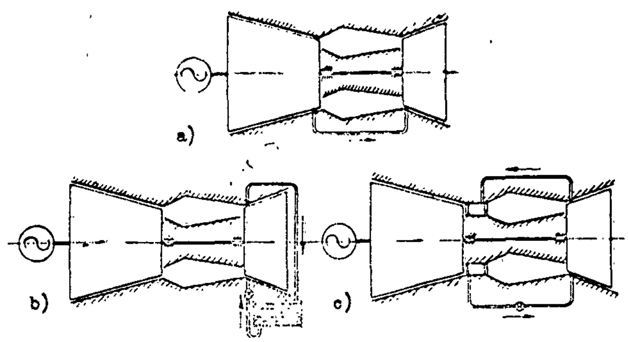

Fig. 1. Cooling systems of gas turbines. a) open system of air cooling; b) closed cooling system with external radiator; c) closed cooling system with heat regeneration.

The calculation of the heat transfer in the working-blade cascades of a rotating turbine wheel from data obtained by blowing through flat cascades is approximate. During the rotation a certain increase in the intensity of the heat transfer from the gas may be observed; this was noted by Ainley [5], who states that the heat-transfer intensity in a turbine wheel increased by approximately 20% in comparison with the heat-transfer intensity in a flat cascade. Apparently, the value of the correction for rotation depends on many factors and can hardly be the same in every case. In order to determine this value more accurately, it is necessary to have experimental data.

Averaged values of the heat-transfer coefficients obviously characterize only the total heat removal from the blades. The distribution of heat fluxes over a blade depends on the nature of the field of values of the heat-transfer coefficients on the streamlined surface,

this field being, as is known, extremely irregular. This irregularity has a significant effect on the temperature field in the blade cross section. Therefore, in order for the blade to cool uniformly, it is necessary to shape the cooling channels appropriately and to distribute them over the cross section.

Let us consider some of the cooling systems already known. In the simplest system of air-cooling a turbine rotor only the disks and interlocking portions of the blades are cooled. In this case a certain decrease in the temperature of the blade quill, owing to heat conduction of the material, is observed. Air-cooling of turbine disks is widely used in gas-turbine construction. The efficiency of this cooling system may be increased by an appropriate organization of the air stream, e.g., by blowing air through the blade joints interlocking with the disk.*

In the case of this type of rotor cooling the strength of the working blades depends essentially on the nature of the temperature field of the gas ahead of the turbine. The gas-temperature distribution around the circumference of the wheel has little effect on the temperature of the working blades, since a natural averaging of the temperature occurs during the rotation of the wheel; the gas-temperature distribution along the radius, on the other hand, directly determines the temperature field in the blade quill and, consequently, the safety coefficients of the quill. Obviously, the problem of finding the optimum laws of distribution of the gas temperature along the radius can be posed. The temperature of a flow which is stagnant with respect to its relative velocity on the working blades of the turbine determines

* Cf., e.g, article by I. T. Shvets and Ye. P. Dyban elsewhere in this issue.

the temperature of these blades and is related to the temperature of the gas ahead of the turbine by the well-known equation

$$T^*_{g_w} = T^*_{g_0} - A\,\frac{c_1^2 - w_1^2}{2g\,c_p}.$$

For the following analysis it is more convenient to assume that the gas-temperature distribution on the working blades $T^*_{g_w}$ is known along the radius. In this case the gas-temperature distribution ahead of the turbine stage T^*_0 may be determined from the dependence given above. Let the gas-temperature distribution along the radius be described by an equation of a quadratic parabola

$$T^*_{g_w} = a + b\bar{x} + c\bar{x}^2,$$

where a, b, and c are constants, and x is a relative coordinate of the cross section $\bar{x} = \frac{x}{h_b}$.

The values characteristic of the gas-temperature field will be

$\Delta T^*_{g_w}$ is the difference between the maximum and the minimum gas temperature;

$\bar{T}^*_{g_w}$ is the average gas temperature;

$\bar{x}_m$ is the relative coordinate of the location of the maximum gas temperature.

By average gas temperature we are to understand in the given case not the average-mass temperature, but a value averaged over the radius, since an accurate calculation shows that this will introduce only a small error (a few degrees).

Assuming that $\Delta T^*_{g_w}$, $\bar{T}^*_{g_w}$, and $\bar{x}_m$ are known, let us determine the constants a, b, and c from the relationships

$$\int_0^1 T^*_{g_w}\,d\bar{x} = \bar{T}^*_{g_w}, \qquad \left(\frac{dT^*_{g_w}}{d\bar{x}}\right)_{\bar{x}=\bar{x}_m} = 0, \qquad T^*_{g_w}(\bar{x}_m) - T^*_{g_w}(1) = \Delta T^*_g, \text{ when } 0 \leqslant \bar{x}_m \leqslant 0{,}5,$$

$$T^*_{g_w}(\bar{x}_m) - T^*_{g_w}(0) = \Delta T^*_g, \text{ when } 0{,}5 \leqslant \bar{x}_m < 1$$

and let us plot the corresponding graphs for the different values of x_m. The graphs are shown in Fig. 2a. It can be seen that the location of the maximum gas temperature has a significant effect on the gas temperature at the root of the blade. The gas temperature at the root in the case of a parabolic law of distribution will be minimum when $\bar{x}_w = \frac{2}{3}$, while the gas temperature at the apex of the blade is equal to the average gas temperature.

Figure 2b shows the change in the safety factor along the blade in the case of linear variation of the cross-sectional area of the quill for various laws of distribution of the gas temperature along the radius. To simplify the calculations it is assumed that the temperature of the blade equals the temperature of the gas. The minimum values of the safety factor are joined by a broken line. As was to be expected, the location of the maximum gas temperature has a very great effect on the minimum safety factor of the quill ($n_{B_{min}}$). The maximum value of $n_{B_{min}}$ will occur when $\bar{x}_m = 1$. However, taking into account that when passing from $\bar{x}_m = \frac{2}{3}$ to $\bar{x}_m = 1$ the minimum safety factor does not increase greatly, while at the same time it is always desirable to have a certain decrease in the gas temperature at the wall of the housing, we must assume that the region of expedient value of the coordinate of the maximum gas temperature should be $\frac{2}{3} \leq \bar{x}_m \leq 1$.

Figure 2d shows the effect of heat removal from the quill into the lock of the blade on the safety factor of the quill in the case of a parabolic law of distribution of the gas temperature along the radius with the maximum temperature being located at the point $\bar{x}_m = \frac{2}{3}$. The cross-sectional areas of the quill were chosen in such a way that the safety factor remained approximately constant over as large a section of the quill as possible. The effect of the removal of heat on the

temperature of the blade is shown in Fig. 2c ($\Delta T^{*'}_{g_w}$ denotes the difference between the maximum and the average gas temperature). It is noteworthy than when the blade lock is cooled the minimum safety factor at first increases appreciably, but then varies slightly with further increase in the quantity of heat removed. This is due to the fact that the cooling of the butt end of the blade propagates only over a small section of the quill (Fig. 2c). In order to cool the entire blade quill, it is necessary to use developed systems of air or liquid cooling.

A whole complex of problems arises in the calculation and designing of turbines with cooling of the blades, disks, housings, and other elements of the turbines: it is necessary to choose the most rational cooling system for the given turbine, to calculate the temperature of the parts being cooled, to develop a design for the turbine to be cooled that is sufficiently simple to manufacture and convenient to operate, etc. The efficiency of a cooling system is generally estimated by calculating the quantity of heat removed during cooling of the heat and the losses introduced by the cooling.

In order to calculate the losses occurring during air cooling, the flow rates of the cooling air over the elements are determined as functions of the admissible temperature of the elements. The calculation of the temperature fields of the turbine blades being cooled is based on the solution of heat-conduction problems with boundary conditions of the third kind. Serving as these conditions are data concerning heat transfer in the cascades from the direction of the gas and in the cooling channels from the direction of the coolant. The solution of one-dimensional heat-conduction problems is not difficult. In the simplest cases the well-known solutions for a flat wall, a

cylinder, and a rod can be used. In more general cases, as was shown by Tirskiy [4], an approximate solution may be obtained by the method of the small parameter. The solution of plane heat-conduction problems by the method of electrothermal analogy is effective in the case of blades, disks, and housings [1], as are approximate methods of reducing planar flows to a system of linear ones.

In the case of many cooling systems heat transfer in cooled blade channels is described by the well-known criterional relationships of convective heat transfer in the presence of forced fluid flow in pipes. If necessary, the local or the over-all intensification of heat transfer in cooling channels may also be otained by methods well-known in heat engineering.

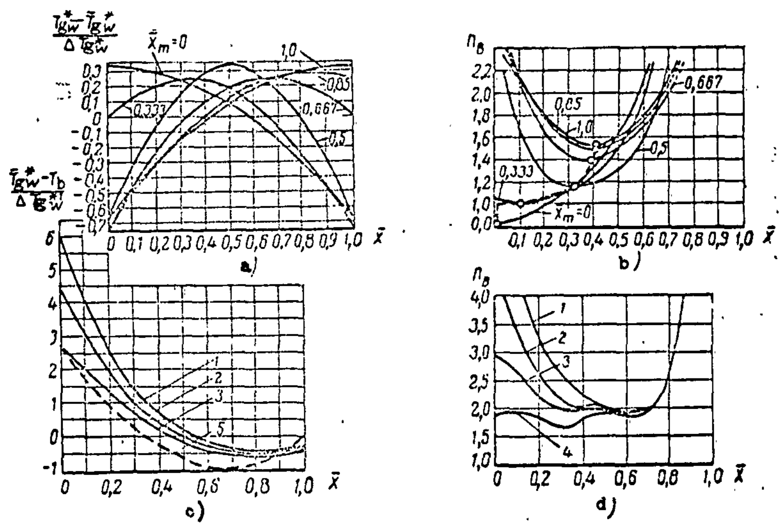

Fig. 2. Cooling of blades by removing heat into the lock: 1) $q_0 = 12 \cdot 10^5$ kcal/m² · hr; 2) $q_0 = 8 \cdot 10^5$; 3) $q_0 = 4 \cdot 10^5$; 4) $q = 0$; 5) $T^*_{g_w}$

Often liquid cooling of a rotor is accomplished by a thermosyphon system, i.e., the effect of natural circulation of a fluid in dead-end cooling channels in a centrifugal-force field is used. In channels

located in the blade quill the feeding of heat to the fluid is accomplished from the lateral surface of the channels, while in channels located in the lock and in the disk the heat is supplied mainly from the end. Obviously, under different conditions of heat supply and heat removal from the fluid the fluid-flow systems in the channel will be different. Therefore the processes of heat and mass transfer in the fluids will also develop differently. Heat and mass transfer in the presence of natural convection of a fluid in a field of centrifugal and gravitational forces has been studied by a number of Soviet and foreign authors [2, 3, 6, 7, and others]. Tirskiy [4] showed that the results of experiments are in satisfactory agreement with the results of theoretical calculations according to formulas obtained from solutions of integral boundary-layer equations. In the case of laminar flow the formula has the form

$$Nu = 0{,}676 \frac{1}{\varepsilon} (CrPr)^{0.25},$$

where ε is a coefficient depending on the physical properties of the fluid Pr and the relative thickness of the boundary layer $\bar{\delta}$.

In the case of a turbulent layer $Nu = 0.0192\ (CrPr)^{0.4}$. Thus the heat transfer in the channels of the blade quill in a centrifugal-force field with natural circulation of the fluid in these channels can be calculated from the formulas cited. In the case where heat is supplied to the fluid from the end of the channel the theoretical work of G. A. Ostroumov [2] and the experimental work of Martin [7] can be used for calculating the heat transfer.

Let us now determine the quantity of heat which must be removed from the turbine during cooling and the losses arising as a result. It is very convenient to use for the analysis of the cooling a dimensionless quantity determined as the ratio between the quantity of heat

removed during cooling of the turbine and the total quantity of heat supplied to the gas in the combustion chamber

$$q_{cool} = \frac{Q_{cool}}{Q_1} = \frac{\Sigma \alpha_g F_b (T^*_{g_w} - T_b)}{G_g C_{pg} (T^*_g - T^*_B)} .$$

This quantity varies only slightly when the absolute dimensions of the turbine are changed, provided that the gasdynamic parameters and the relative dimensions of the cascades are chosen within the limits of the values used in present-day turbines. Actually the quantity q_{cool} may be written in the form

$$q_{cool} = \frac{K \cdot \frac{b}{t}}{\sin\beta} \cdot \frac{C}{Pr\, Re^{1-n}} \cdot \frac{T^*_{g_w} - T_g}{T^*_g - T^*_B},$$

if, instead of α_g and F_b, we substitute

$$\alpha_g = \frac{C\, Re^n \lambda_g}{b}, \qquad F_b = z_b \cdot \Pi \cdot h_b = \pi D_T h_b \frac{\Pi}{t},$$

where Π is the perimeter of the profile of the blade cross section; its value may be assumed to be proportional to the chord of the profile $\Pi = k \cdot b$;

C is a constant in the equation $Nu = CRe^{\Pi}$;

Pr is the Prandtl number;

$\pi D_T h_b$ is the axial area of an annular cascade, $\pi D_T h_b = \frac{G_r}{w \gamma \sin\beta}$;

β is the angle between the velocity vector and the plane of rotation of the wheel (for the nozzle cascades this angle equals α_1).

When the dimensions of the turbine change, the Re number changes; when the dimensions decrease, the Re number decreases. However, since the degree of n is large (n = 0.7-0.8), a severalfold change in the Re number does not lead to any great changes in the value of q_{cool}. For example, if when n = 0.75 the Re number is reduced by a factor of three, which corresponds, at the very least, to a threefold change in the dimensions, the value of q_{cool} changes by a factor of approximately

1.3.

The cooling depth $(T^*_{g_w} - T_b)$ has a great effect on the relative heat removal and depends on the properties of the material, the acting loads, and the gas temperature. The maximum tensile stresses in the root cross section of the quill of a working blade are determined by the formula

$$\sigma_p = \frac{2\gamma_m}{g}\frac{u_T^2}{\Theta}\psi,$$

where γ_m is the gravimetric density of the blade material;

ψ is a coefficient which takes into account the decrease in the stresses at the root as a result of conicity of the blades; in the claculations it can be regarded as constant and equal to $\simeq$ 0.506 for $\frac{f_\Pi}{f_c} = 0.25$;

θ is the ratio between the average diameter of the turbine and the length of the blade; in subsequent calculations this quantity is assumed to be equal to 3.5 for the last stages of all the turbines;

u_T is the peripheral velocity of rotation of the rotor; it was determined by choosing $\frac{u_T}{C_{ad}}$ in the last stage, the temperature drop in this stage, and the degree of reactivity. It was assumed that the over-all temperature drop in the turbine is distributed uniformly over the stages.

Thus the over-all temperature drop in the turbine and the stresses at the roots of the working blades of the last stages can be determined for every value of the gas temperature ahead of the turbine and every value of the degree of reactivity of the air pressure in the compressor. In all the preceding stages of a single-shaft multistage turbine the stresses will vary in proportion to the change in the axial area of the annular cascades.

In determining the admissible temperatures of the working blades the safety factor n_b was taken equal to 2 and it was assumed that the turbine blades are made out of EI-437B alloy. The temperature of the nozzle blades was taken to be 850°C. The gas temperature on the nozzle and working blades $T^*_{g_c}$ and $T^*_{g_w}$ and the air temperature beyond the compressor were determined from the usual formulas.

In Fig. 3 the solid lines indicate the results of a calculation of the relative heat removal during cooling of the turbines in the coordinates (q_{cool}, T^*_g); on the left for $\Pi_c = 6$; on the right for $\Pi_c = 18$. This calculation was performed for various stage loads, thus corresponding to choosing different numbers of stages for each gas temperature ahead of the turbine. The broken lines correspond to the results of a calculation of the heat removal at a constant temperature of the working blades. As was explained, the graphs comprise a wide class of turbines with respect to dimensions and number of stages. It goes without saying than when the results of the calculations are extrapolated into the region of very small or large turbine dimensions we have to reckon with the effect of a change in the Re number on the relative heat removal.

It can be seen from the graphs that the relative heat removal in the case of moderate stage loads and high gas temperatures is an appreciable amount. For example, when $T^*_g = 1800°K$, $H_{wall} = 15$ kcal/kg, $q_{cool} \cong 15\%$, in which case a change in Π_c from 6 to 18, for all pracitcal purposes, has no effect on the value of q_{cool}. An increase in the stage load appreciably decreases the heat removal during cooling but is accompanied by an increase in the stresses in the blades, if the ratio $\frac{D_T}{h_b}$ is kept constant.

It is interesting to consider how the heat removal will vary, if

the increase in the peripheral velocity required for an increase in the stage load is realized by increasing the diameter of the wheel, and not the number of rotations of the rotor. In this case the stresses in the working blades will not increase.

The results of such a calculation of heat removal from the turbine wheel for $T_g^* = 1800°K$ and $\pi_c = 18$ are shown in Fig. 4 on the left. When the stage load is increased, the heat removal is reduced more intensively in a turbine in which an increase in the peripheral velocity is achieved by increasing the diameter of the wheel while keeping the number of rotations of the rotor and, consequently, the stresses in the working blades constant. This facilitates the manufacture of the cooled blades, since their length decreases in proportion to an increase in the diameter.

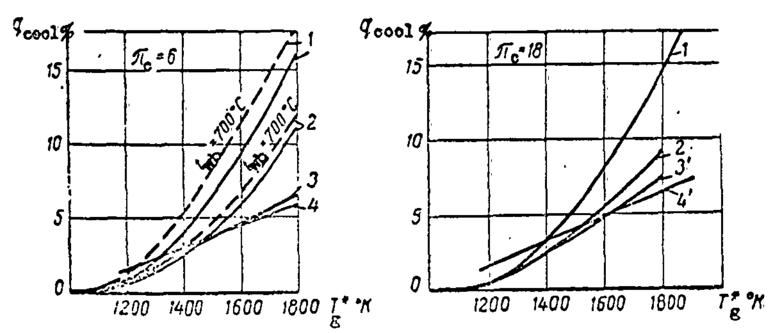

Fig. 3. Heat removal during cooling of turbines: 1) H_{wall} = 15 kcal/kg; 2) H_{wall} = = 25; 3) H_{wall} = 42; 4) H_{wall} = 44.3; 3') H_{wall} = 59; — — — — at the same temperature for the cooled working blades of all the stages t_{wb} = 700°C.

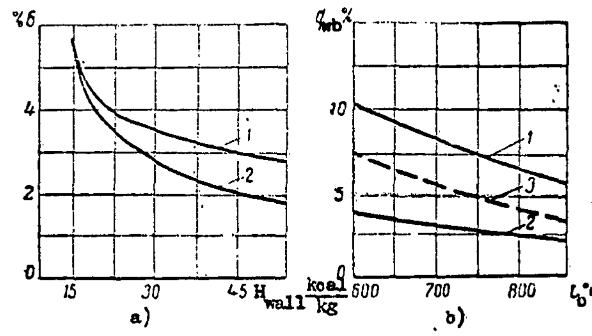

Fig. 4. Decrease in heat removal from gas during cooling of turbines: a) $\pi_c = 18$, $T_g^* = 1800°K$, 1) θ = const, 2) σ_p = const; b) $\Pi_c = 18$; 1) $T_g^* = 1800°K$, $H_{wall} = 15$ kcal/kg; 2) $T_g^* = 1800°K$, $H_{wall} = 42$ kcal/kg; 3) $T_g^* = 1600°K$, $H_{wall} = 15$ kcal/kg.

The heat removal also decreases during cooling when the temperature of the blades increases (Fig. 4, right). However, this involves the development of more highly heat-resistant alloys for turbine blades, thereby giving rise to certain difficulties.

The removal of heat from the gas during its expansion in the turbine is, of course, accompanied by a decrease in the work of the turbine and its efficiency, even if we assume that the hydraulic losses in the cooled cascades are the same as those in the uncooled ones*. The decrease in the work of the turbine during heat removal from the gas can be determined from the equation of conservation of energy. If we assume that the gas velocity at the outlet of a cooled and an uncooled turbine is the same, then

$$A L_T - A L_{T\,cool} = Q_{cool} - (i_4 - i_{4\,cool}).$$

The areas equivalent to the values of Q_{cool} (single hatching) and

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* There is justification for such an assumption (cf., e.g., V. T. Ivanov).

$(AL_T - AL_{cool})$ (double hatching) are shown conventionally in a TS diagram in Fig. 5.

The area S_Δ (area 3 - 4 - 4_{cool}) may be determined approximately from the formula

$$S_\Delta = \frac{\ln \frac{T_4}{T_{4cool}}}{\ln \frac{T_4}{T_{4ad.}}} \alpha_\infty \cdot H_T ,$$

where α_∞ is the heat-recovery factor for an infinite number of stages;

H_T is the available temperature drop in the turbine.

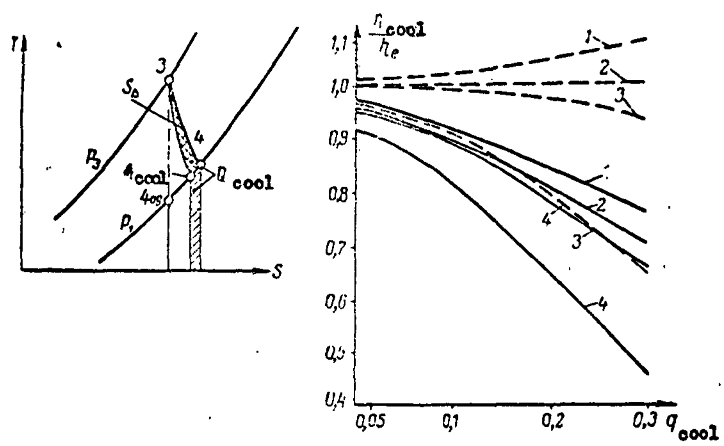

Fig. 5. Losses in fluid cooling of turbines:
——— cooling without regeneration of heat;
— — — regeneration of cooling heat.
1) π_c = 6, T_g^* = 1800°K; 2) π_c = 18, T_g^* = 1800°K;
3) π_c = 6, T_g^* = 1200°K; 4) π_c = 18, T_g^* = 1200°K.

The actual efficiency of a cooled turbine will be less than the efficiency of an uncooled turbine by the amount

$$\Delta = \frac{\ln \frac{T_4}{T_{4cool}}}{\ln \frac{T_4}{T_{4ad}}} \alpha_\infty .$$

that is

$$\eta_{T_{cool}} = \eta_T - \Delta .$$

Using the parameters introduced, q_{cool} and Δ, it is not difficult to obtain the formulas for calculating the actual efficiency of a cycle with a cooled turbine. If the heat removed from the gas during the cooling of the turbine does not return to the cycle, e.g., is dissipated in an external air stream, then

$$\eta_{e_{cool}} = \eta_e \left(1 - \frac{\Delta}{\varphi} \right),$$

where

$$\varphi = \frac{H_T \eta_T - \frac{H_c}{\eta_c}}{H_T}.$$

In the case of regeneration of the cooling heat according to the system depicted in Fig. 1c the efficiency of the cycle will be

$$\eta_{e_{p\,o}} = \eta_e \frac{1 - \frac{\Delta}{\varphi}}{1 - q_{cool}}.$$

In both formulas η_e denotes the actual efficiency of the cycle of a machine with an uncooled turbine.

Calculations show that the value of Δ is proportional to q_{cool} and, on the average, can be taken equal to

$$\Delta = (0{,}3 \div 0{,}5)\, q_{cool}$$

Figure 5, right, shows how heat removal from the gas during the cooling of the turbine affects the efficiency of the cycle. The calculation was made according to the formulas given above, it being assumed that $\Delta = 0.45\ q_{cool}$. As can be seen, heat removal from the gas has an appreciable effect on the efficiency of the cycle.

It was shown above that at the gas temperature $T_g^* = 1800°K$ the heat removal during the cooling of the turbine may amount to 15% of the total quantity of heat supplied to the cycle. If measures to regenerate the cooling heat are not taken in this case, this heat

removal will reduce the efficiency of the machine by approximately 10%. In the case of complete regeneration of the cooling heat the efficiency of the cycle increases to values which almost coincide with the efficiency of the machine cycle without cooling.

Such are the over-all thermodynamic properties of closed cooling systems. In turbines with open systems of air cooling the conversion of gas energy into work is distinguished by a number of special features. On the one hand, there is no external heat removal; the heat removed from the blades returns to the stream together with air. On the other hand, the mixing of the gas stream with the air stream causes irreversible losses in the kinetic energy of the streams, as a result of which the total pressure in the stream beyond the cascade being cooled decreases. A certain part of the turbine power is spent in pumping air through the cooling system. The power and efficiency losses of the engine in the case of air cooling of a turbine will obviously be proportional to the relative consumption of air required for cooling. Approximate equations for determining the losses may be obtained from the equation of conservation of energy written for the gas and air streams separately, since the return of the cooling heat to the gas stream containing heated air enables us to consider separately the processes of expansion of the gas and the air in the turbine.

The processes of compression and expansion of the cooling air are shown conventionally in a T-S diagram in Fig. 6, left. The hatched areas are equivalent to the work which must be expended in order to realize the processes. When the air required for cooling is taken from the intermediate stages of the compressor, the amount of this work decreases. In the case of air cooling of the turbine the engine

power will decrease as a result of the removal of part of the air for the purpose of cooling and the expenditure of work in pumping the cooling air through, while the efficiency of the engine will decrease only as a result of the expenditure of work in pumping air through. Since both quantities are proportional to the relative consumption of cooling air, the relative changes in the power and specific flow rates of fuel may be written in the form

$$\Delta \overline{N}_c = \frac{N_c - N_{c\,cool}}{N_c} = \frac{G_B}{G_g}(1+\bar{f}),$$

$$\Delta \overline{C}_c = \frac{C_{c\,cool} - C_c}{C_c} = \frac{G_B}{G_g}\cdot \bar{f},$$

where

$$\bar{f} = \frac{L_0}{L_c}\left(1 - \eta_K \cdot \eta_p \cdot \frac{T_2}{T_{2ad}}\right)$$

L_c is the work of compressing the air;

L_e is the useful work of the cycle;

η_c is the efficiency of the air-compression process and is assumed to be equal to the efficiency of the compressor;

η_p is the efficiency of the air-expansion process in the turbine.

The later value may be determined approximately by successive calculation of the turbine stages, allowing for expulsion of air from the cooled cascades, and may be determined more exactly from tests of cooled turbines.

Figure 6, right, shows the values of the coefficient $\bar{f}$ as a function of the expansion efficiency. The average values of the expansion efficiency, as is shown by calculations, lie within the limits η_p = 0.6-0.8, for which $\bar{f}$ = 0.3-0.6.

Thus it may be assumed that in the case of air cooling of a turbine engine a 1% air consumption for cooling causes, on the average, a 1.5%

decrease in the engine power and a 0.5% increase in the fuel consumption.

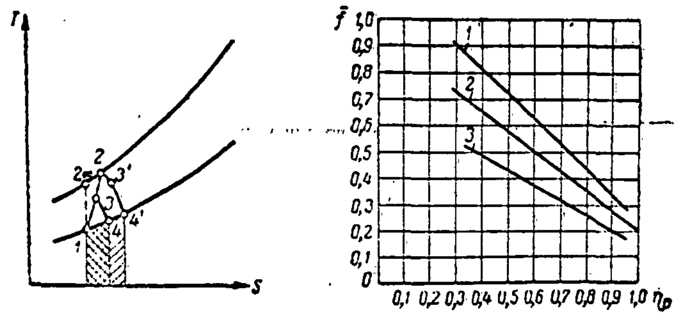

Fig. 6. Losses resulting from air-cooling of turbines: $T_g^* = 1300°K$; $\pi_c = 7$, $\eta_c = 0.8$, $\eta_p = 0.85$; 1) degree of increase in air pressure $\pi_c = 7$; 2) $\pi_c = 5$; 3) $\pi_c = 3.5$.

The absolute values of the flow rates of cooling air are determined by simultaneous solution of the heat-balance and heat-conduction equations. By such calculations it was established that the flow rates of air for cooling the above-mentioned turbines increase appreciably when the gas temperature increases, and at about 1400°K the total consumption is more than 10% of the gas consumption. It is clear that such an increase in the consumption of cooling air makes it inadvisable to use air cooling for turbines at gas temperatures above 1400°K. In practice, it may prove to be advisable to combine air and liquid cooling of turbines. For example, in the case of air cooling of the nozzle apparatus alone and liquid cooling of the rotor moderate air flow rates at comparatively high gas temperatures are obtained.

In conclusion, we can state the following brief résumé:

At the present time there is a sufficiently complete substantiation of the possibility of creating efficient cooling systems for

turbines. The choice of the specific cooling system depends on the temperature level and on the specific purpose for which the gas-turbine apparatus is intended. From an approximate analysis of the efficiency of various cooling systems of multistage turbines it can be seen that air cooling according to an open system should be used at gas temperatures no higher than 1400°K and with large temperature drops on the stage. Liquid cooling of turbines can, in practice, ensure their efficiency at very high gas temperatures. However, the losses resulting from cooling may be considerable. In order to decrease the losses, it is desirable to use regenerative cooling systems. The heat removal in multistage cooled turbines may be appreciably decreased by increasing the temperature drops on the first stages, in which case, in order to maintain optimum ratio $\frac{u_T}{C_{ad}}$, it is more advantageous to increase the diameters of the turbine wheels, thereby intensifying the reduction in the heat removal and facilitating the creation of cooled blades as a result of a decrease in the length of the latter.

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A STUDY OF TRANSIENT AERODYNAMIC PHENOMENA IN AXIAL COMPRESSORS

G. S. Samoylovich, Docent and Candidate of Tech. Sciences

(Moscow Power-Engineering Institute)

The problem of transient aerodynamic phenomena in turbomachines (rotating stall, stall flutter, surge, etc.) is considered in this article. Certain results obtained at the Department of Steam and Gas Turbines of the Moscow Power-Engineering Institute are communicated.

A study of transient aerodynamic phenomena in gas turbines and axial compressors is of great practical interest, since it enables us to determine the possible range of operation of gas-turbine assemblies, to study the starting regimes, and to determine the dynamic stresses in the blades.

A number of phenomena occurring in turbines and compressors can be explained only by considering unsteady flows. Such phenomena include surge, rotating stall, stall flutter, and aerodynamic damping. Some of them have been discovered comparatively recently, and, although the theoretical and experimental investigation of them is far from complete, it is clear that they must be taken into account in the

designing and calculation of gas-turbine assemblies. A study of these unsteady aerodynamic phenomena in turbomachines can give valuable information required for the choice of the assembly system, the designing of the control, and the calculation of the individual units.

Unsteady aerodynamic phenomena interest us mainly from the standpoint of reliable operation of the blading of turbomachines. It is known that dynamic stresses arising in the blades during vibration may considerably exceed the static stresses for which they were designed. In the present article we are interested only in the aerodynamic causes of the excitation of blade vibrations.

At least four characteristic phenomena can be distinguished:

a) forced blade oscillations arising under the action of periodic aerodynamic forces caused by nonuniformity of the flow. Consequently, these oscillations are maintained by a periodic source of energy;

b) self-excited oscillations of blades in a uniform gas stream. Self-excited oscillations are oscillations maintained by an aperiodic source of energy;

c) blade oscillations caused by the self-exciting, unsteady asymmetry of the flow. In this case, in contrast to case b), there arises a gas-stream self-oscillation, which exists independently of the blade oscillations or, to put it in a better way, a blade vibration arises as a result of self-oscillation of the flow;

d) blade vibrations caused by self-oscillations of the flow in the gas or air lines.

This classification is, of course, not complete, but as of the present time it seems clear that these phenomena are, for all practical purposes, more important than the others.

When we consider the causes of blade oscillations, we note that

they can be divided into two types. To problems of the first type belong those which are described only by aerodynamic equations. To the second type belong those problems in which elasticity of the blades plays a fundamental role. To put it more strictly, it can be said that in problems of the first type the dependence of the blade deformation on aerodynamic conditions is essential, while the dependence of the aerodynamics of the flow on the deformation of the blades is unessential.

Problems of the second type, on the other hand, are characterized precisely by the presence of an essential inverse relationship between the deformation of the blades and the aerodynamics of the flow.

In this article we shall discuss some of the work being done at the Moscow Power-Engineering Institute in the study of unsteady aerodynamic phenomena in turbomachines. The article deals only with experimental studies of the aerodynamic factors b) and c) enumerated above.

Before speaking of the results of the experiments, let us briefly describe the measuring apparatus, since the results obtained always depend on the apparatus used (this applies to an even greater degree to studies of unsteady processes).

For a study of the aerodynamics of rotating stall, surge, etc. it is necessary to have low-inertia probes suitable for measuring velocity pulsations, static pressure, and other quantities. The basic technical requirements of low-inertia probes to be used in such a study should be as follows:

a) the natural frequency of the probe (we have in mind the natural frequency of the system consisting of the receiving, transmitting, and recording apparatus) should be sufficiently high, so as

not to distort the actual picture of the unsteady flow;

b) the over-all dimensions of the probes should be of minimum size, so that they can be used for measurements in the flow area in the axial gap between the rows of cascades;

c) it is desirable that the chosen principle of measurement be such that on the basis of it probes can be created for measuring the total pressure, the static pressure, and the velocity and directional angle of the flow (naturally we have in mind the measurement of these quantities as functions of time);

d) the schemes should allow simultaneous recording of the readings of many probes, thus making it possible to ascertain the time relationship between the flow parameters.

In our work we designed, constructed, and thoroughly tested probes which, to a sufficient extent, satisfy the above requirements; they may be called tensometric, since their sensitive element in an elastic plate of circular or rectangular shape with a tensometric pickup. The design of these probes varied greatly. We give as an example a photograph of the total-pressure probes (Fig. 1). In these probes the thickness of the elastic plate is equal to 0.03-0.15 mm, while the pickup is wound out of constantan wire with a diameter of 0.03 mm in the form of a spiral with a diameter of 4 mm.

The plate is located in the head of the probe. The head has a thickness of only 2-2.5 mm. The probes shown in the photograph were used for measuring pulsations in full-scale compressors of gas-turbine assemblies (GT-12, of the Leningrad Metal-Working Plant). They have, as can be seen from the photograph, a cylindrical chamber, which is connected with a space under a membrane and, through a special choking device, has an outlet into the flow. Thus the membrane is kept in

equilibrium by the average pressure and receives only the pulsation component of the total pressure.

Let us now dwell on the results of the study of rotating stall. The phenomenon of rotating stall consists of the following. Let us assume that at a certain air flow rate an axial-compressor stage operates normally, i.e., no stalling is observed. We shall now somehow decrease the volume flow through this stage. This will lead to an increase in the angle of attack. It seems that if the angle of attack during cascade flow approaches the value at which blade stall should occur, stalling is observed not on all the blades of the cascade simultaneously, as might be expected, but only on a certain group of blades (or several groups). These stalling zones do not remain stationary, but move. Therefore the phenomena is called a moving or rotating stall.

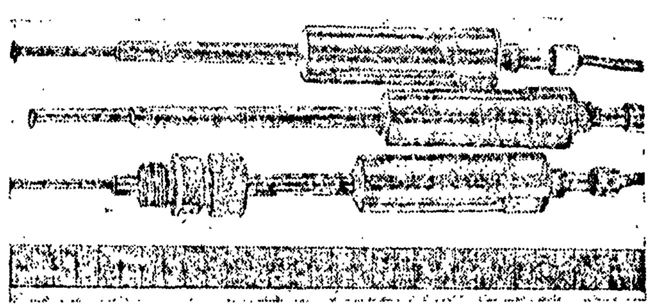

Fig. 1. Low-inertia tensometric total-pressure probes.

Let us consider the experimental characteristic of the axial-compressor stage (Fig. 2).

The compressor characteristic plotted indimensionless coordinates is $\varphi = \frac{ca}{u}$, $\psi = \frac{\Delta p}{\rho u^2}$. This characteristic was recorded for a wide range of change in the flow coefficient down to $\varphi = 0$ with a thorough study being made of the conditions in the stalling zone.

Several characteristic ranges of operation of compressors may be noted. The first range of normal operation is $\varphi > 0.23$. If the flow rate is decreased with the number of rotations remaining constant, so that the outlet resistance is increased, at first ψ will increase (up to about $\psi = 0.4$), and then the operating conditions of the compressor will change abruptly. The flow coefficient will decrease to about $\varphi = 0.13$, while the pressure coefficient will decrease to $\psi = 0.20$. This decrease in the coefficients φ and ψ is caused by a rotating stall. With further decrease in the flow rate the coefficient φ continues to decrease down to $\psi = 0.17$ (second range), is then stabilized, and even increases somewhat (third range). If the flow rate is now increased with the number of rotations remaining constant, thus decreasing the aerodynamic resistance at the outlet, the third and second ranges will be transversed in inverse order, but then, instead of operating in the first range, the compressor will begin to operate in a fourth range.

This range is characterized by the fact that it is realized only during so-called "reverse running." The white experimental dots in Fig. 3 were obtained during a decrease in the flow rate, while the black dots were obtained during an increase. If, while increasing, the flow coefficient reaches $\varphi = 0.23$, then an abrupt change in the operating conditions of the stage occurs again, the rotating stall disappears, and the compressor passes over to normal operation. This behavior of the external characteristic is related to the processes occurring in the blades.

Let us consider the results of a study of the flow in the flow area with the aid of low-inertia total-pressure probes. Figure 3 contains oscillograms recorded for the four characteristic ranges. These oscillograms are a recording of the readings of two low-inertia

total-pressure probes located ahead of the impeller. The first oscillogram of this series refers to the range of normal operation of the compressor ($\varphi = 0.24$), and therefore the readings of the probes are recorded in the form of two equal straight lines indicating that the cascade-flow regime is steady. The third oscillogram refers to the second range of operation ($\varphi = 0.13$). Note especially that the curve of the readings of each probe is periodic and consists of straight-line segments alternating with sections of strong pulsations. Obviously, wherever the probes write a straight line, the working cascade operates in the normal regime, while wherever pulsations appear, the flow around it causes stalling. Note that the curves written by the two probes are identical, but shifted in phase, thus indicating a rotation of the stalling zone, since the probes are set at an angle to each other.

In the case of complete cessation of the flow ($\varphi = 0$) the stalling encompasses the entire working cascade, while the probes indicate the presence of strong pulsations.

Finally, oscillogram 3b was recorded during the operation of the compressor stage in the fourth zone. This oscillogram shows even more clearly than the preceding ones the boundaries between the stalling and nonstalling regions. As might be expected, the relative magnitude of the stalling zone decreased.

From the oscillograms we can calculate the relative arc length occupied by the stall $\underline{e}$ (in relation to the length of the circumference), as well as the rotational velocity of the stall and u_{st} in relation to the peripheral velocity of the blades. The corresponding graphs are shown in Figs. 4 and 5.

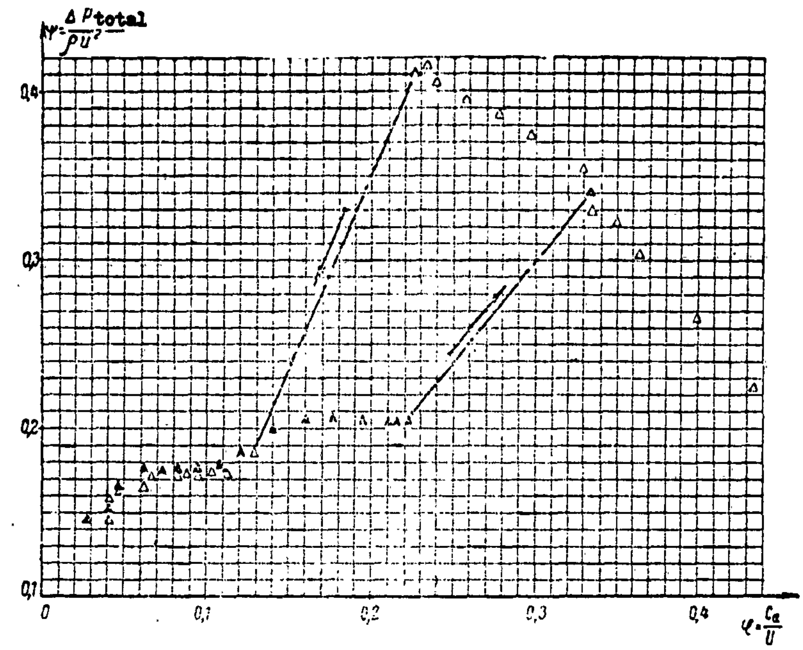

Fig. 2. Experimental characteristic of an axial-compressor stage.

The length of the stalling zone depends almost linearly on the flow coefficient, while the ratio between the rotational velocity of the stall and the peripheral velocity of the blades is equal to approximately 0.43 (for the fourth zone of operation) and decreases somewhat at small φ.

An analysis of the tests shows that in the given case there exists only one stalling zone, which immediately encompasses the entire height of the blade. In these tests we recorded simultaneously both the change in the static pressure in time and the flow velocity.

Of course, these parameters are also variable. By way of illustration, Fig. 6 shows an oscillographic recording of the flow velocity ahead of the stage. The recording of the velocity in the given case was done with the aid of a thermoanemometer, which sends a signal to the screen of a cathode oscillograph with a periodic sweep and an afterglow, thus making it possible to photograph the press.

Considering the results of the experiments, we can draw certain conclusions concerning the processes occurring in the stage.

When the stage is operating in the stalling regime, the entire cascade is divided into two regions: a region with the relative area $1 - e$, in which there is no stalling, and a region with the relative area $\underline{e}$, where the blades are stalling. From the photographs of the recording of the total pressure, the velocity, and the static pressure it can be seen that these regions are outlined fairly distinctly.

Let us attempt to explain the nature of the dependence of the relative area of the stall on the flow coefficient φ. Let Q denote the air flow supplied to the system by the stage, while Q_1 and Q_2 are the flows passing through the nonstalling and stalling regions, respectively.

In the stalling regions the air flow is very small, equal to zero, or even moving in the reverse direction. The possibility of such a reverse flow was observed in the tests and was recorded by the measuring system. The recording of the change in the flow velocity ahead of the impeller may serve as an illustration (Fig. 6). The negative velocities correspond to periodic peaks pointing upward. The problem is that the thermoanemometer probe senses only the modulus of the velocity, but does not react to its direction, and therefore a change in the sign of the velocity corresponds to a turning point of the curve on the oscillogram.

As a result, we can write the flow balance as follows

$$Q = Q_1 - Q_2$$

We may write the following expressions for the flows

$$Q = c_a F, \quad Q_1 = c_{a_1} F(1-e), \quad Q_2 = c_{a_2} F e,$$

where:

F is the annular area at the inlet to the working cascade;

c_a is the nominal axial velocity at the inlet and is calculated from the flow through the system;

c_{a_1} and c_{a_2} are the average axial velocities in the normal zone and the stalling zone, respectively.

The flow rate Q may be represented in terms of the coefficient φ:

$$Q = u \varphi F.$$

where u is the peripheral velocity of the blades.

The flow rate Q_1 may also be represented in terms of the flow coefficient, but taken along the right-hand normal branch of the characteristic with the actual pressure coefficient ψ:

$$Q_1 = K_1 \varphi_1 F(1-e) u.$$

where K_1 is a coefficient which takes into account a certain deterioration in the work of the impeller, owing to a disruption of the normal inleakage of air on the boundary between the zones.

It may be assumed that the velocity c_{a_2} is proportional to $\sqrt{\frac{\Delta p}{\rho}}$, where Δp is the pressure drop on the stage under the given conditions:

$$Q_2 = K_2 \sqrt{\frac{\Delta p}{\rho}} F e.$$

where K_2 is a proportionality coefficient.

In general, K_1 and K_2 depend on the operating conditions, but in a first approximation they may be assumed to be constant.

Substituting these expressions into the flow balance, cancelling out the F's, and dividing by u, we obtain:

$$e = \frac{K_1 \varphi_1 - \varphi}{K_1 \varphi_1 + K_2 \sqrt{\psi}}.$$

If we calculate the coefficients K_1 and K_2 with the aid of the characteristic φ-ψ and the graph e-φ, they are found to be equal to $K_1 = 0.72$ and $K_2 = 0.26$.

The structural formula obtained explains the general nature of the relation between the compressor characteristic and the relative stalling area. On those sections of the characteristic where ψ = const (the static pressure coefficient is also approximately constant there) e depends linearly on φ (since φ_1 = const and ψ = const). On the other hand, where ψ decreases, the increase in c occurs more intensively.

It should be especially emphasized that during the appearance of rotating stall air oscillations were observed only in the impeller blade system. The air masses in the system did not participate in the oscillations, as was corroborated by direct measurements. Consequently this phenomenon should in no way be confused with surge, although a rotating stall may cause surge.

Let us now describe the study of stall flutter arising in a compressor-blade cascade. This type of self-oscillations of blades from classical flutter, which is well known in aviation, in the fact that they do not require two degrees of freedom in order to arise: the blades may undergo only bending oscillations. Purely bending oscillations of blades are extinguished by the flow (aerodynamic damping). However, this conclusion is valid only under the condition that an increase in the angle of attack corresponds to an increase in the lifting force. Such a dependence of the normal force (later we shall speak not of the lifting force, but of the force of the normal chord of the blade, since it is preceisely this force which accomplishes the work during oscillation) is characteristic of the region of separationless cascade flow. At large positive angles of attack there occurs

a separation of the flow from the backs of the blades near the inlet edge in the zone of large positive pressure gradients. We are interested above all in the fact that after the onset of stalling the force characteristic of the blade is reversed, i.e., an increase in the angle of attack corresponds to a decrease in the normal force. We give by way of illustration a graph (Fig. 7) plotted from measurements of the normal force made with the aid of a special three-component tensometric balance which allowed us to weigh the forces and the moment acting on a blade in a cascade.

Flutter, which arises at postcritical flow angles, is characterized by the fact that it exists at comparatively low flow velocities and is therefore dangerous for blades of turbomachines.

Self-oscillations of the stall-flutter type may occur either in a cascade or when the blade is isolated. However, it does not follow from this that the cascade parameters have no effectoon the location of the boundaries where flutter originates, the dynamic stresses in the blades, etc. The cascade effect is realized, above all, through a change in the force characteristic "angle of attack — normal force." Let us assume that the blade undergoes oscillations with a small amplitude in the separation flow regime, which corresponds to the descending portion of the force characteristic. It is easy to show that the blades should receive energy from the flow. This energy will sustain the oscillations of the blade, and, if the amplitude is small, it will increase. It is essential that the amplitude be stabilized after a certain period of time. This maximum amplitude and the corresponding maximum cycle of oscillations are characterized by the energy balance: the sum of the energies supplied to and removed from the oscillating blade is equal to zero. It should be born in mind that

the energy removal is realized both by mechanical and aerodynamic damping.

The aerodynamic processes occurring during stall flutter are very complex. The phenomenon of hysteresis plays a large role in stall flutter. The delay in establishing circulation around an airfoil during a change in the angle of attack is related to the periodic process of separation and adhersion of the boundary layer. Thus the use of a quasi-static characteristic is not entirely legitimate. Such a delay may have a decisive effect, especially when the oscillation frequency of the blade is high.

The mechanism of stall flutter is closely related to the intensity and frequency of the separations of the boundary layer from the blades during flow at postcritical angles of attack. Therefore when studying stall flutter we must not consider time-averaged aerodynamic forces, but must study the pulsation occurring in the lifting force during periodic separations. Periodic separations of the boundary layer and changes in circulation occur, as is known, even on a nonvibrating blade. The frequency and intensity of the separations depend on the geometrical parameters of the cascade and on the flow regime. In the case of separation flow around vibrating blades the frequency and intensity of the separations depend on the frequency and amplitude of the oscillating blades. As was shown by experiments, the frequency of the separations can be tuned to the oscillation frequency of the blade, and frequency magnification, as it is called in radio engineering, is observed.

Moreover, the real flow picture has, of course, a three-dimensional character with the root portion of the blade receiving energy from the flow and the upper portion operating under conditions of aerodynamic damping. More precisely, at the root of the blade the work of the

internal-damping forces of the material is greater than the work of aerodynamic excitation, on the central part of the blade aerodynamic excitation dominates, and, finally, at the apex aerodynamic damping plays the main role.

An experimental study of the hysteresis effect and the distribution of work over the height of a vibrating blade, as well as a study of the frequency characteristic of the boundary-layer separations, is very important for explaining the mechanism of stall flutter and aerodynamic damping. At the same time, a prediction of the zone of existence of stall flutter and an estimate of the dynamic stresses can be made on the basis of integral measurements and the static force characteristic.

A typical oscillogram of the dynamic bending stresses in the root of the blade for an angle of attack greater than the stall angle is shown in Fig. 8. Characteristic is the fact that the stress amplitude remains practically constant over a considerable number of cycles, while the oscillations occur with a frequency practically equal to the natural frequency of the blade in the first tone. These are the main features of a self-oscillation process.

The maximum dynamic stresses in the given flow regime amount to $\pm$ 570 kg/cm^2 for a corresponding static pressure of 340 kg/cm^2.

At the same time, in certain cases it is found that the oscillations are of a sporadic nature not characteristic of a strictly self-oscillation process. A zone of almost constant amplitude alternates with zones where the stresses decrease almost to zero, i.e., during a certain interval of time the oscillations are damped. There are several reasons for this dependence of the stresses on time. First of all, separation flow depends on many random factors and is not stable,

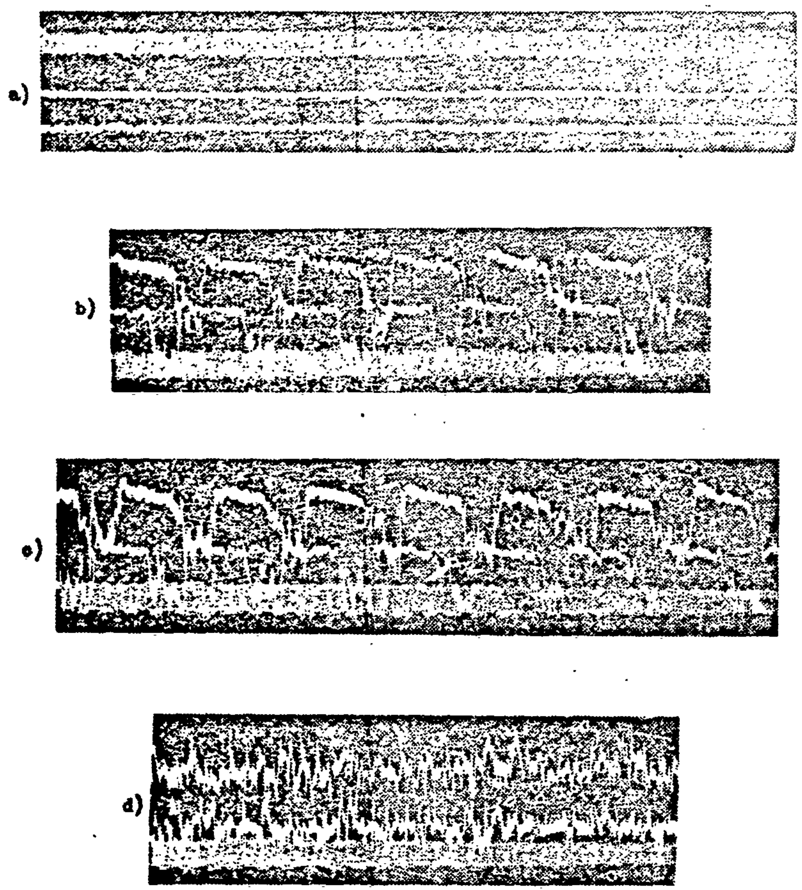

Fig. 3. Oscillographic recording of the total pressure ahead of the impeller.

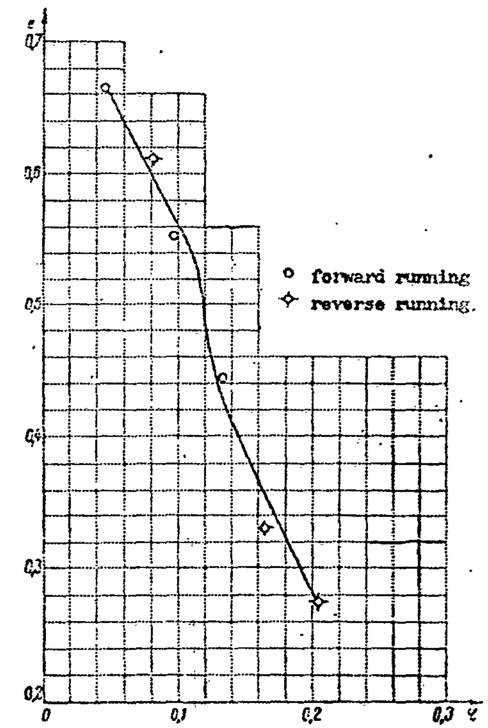

Fig. 4. Relative area occupied by a rotating stall as a function of the flow coefficient.

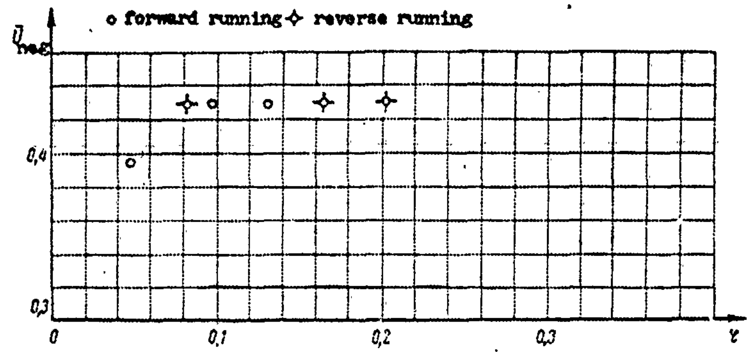

Fig. 5. Dimensionless velocity of a rotating stall as a function of the flow coefficient.

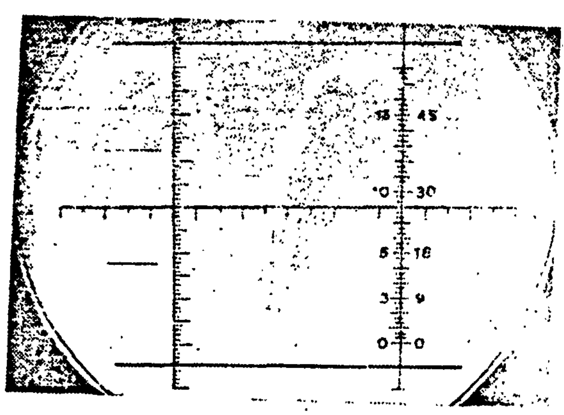

Fig. 6. Oscillographic recording of flow velocity.

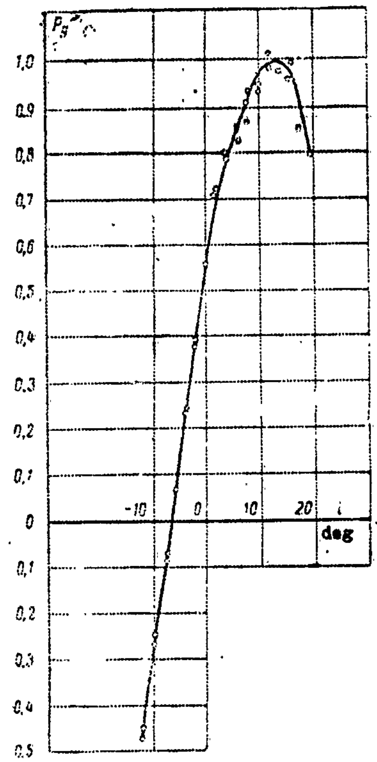

Fig. 7. Dependence of the dimensionless normal force on the angle of attack.

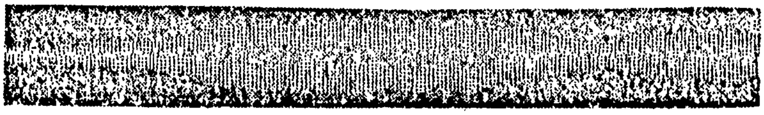

Fig. 8. Oscillographic recording of the dynamic stresses in a blade during stall flutter.

and consequently the zerodynamic reproducibility and the periodicity of the process are disrupted within certain limits. Secondly, the stress depends on the possibility of tuning the vortex-separation frequency to the natural frequency of the blade, thus possibly causing beats. Finally, a moving stall may be superimposed on the stall flutter, thus disrupting the frequency of the phenomenon, since the blades move out of the separation flow.

Figure 9 shows the experimental dependence of the dynamic stresses in vibrating blades on the angle of attack α for 4 (sic) different Mach numbers of the incident flow.

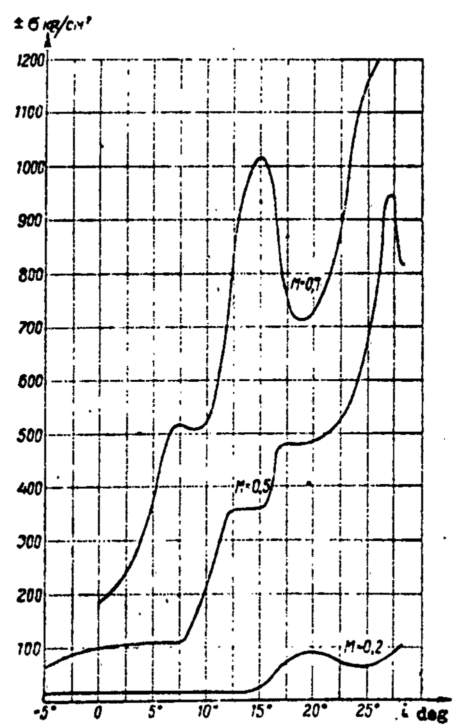

Fig. 9. Dynamic stresses in blades during stall flutter.

With the aid of the same measuring apparatus and investigational procedure thorough tests of full-scale axial compressors of the 12,000 kw gas-turbine assembly produced by the Leningrad Metal-Working Plant were carried out. In certain operating regimes rotating stalls were detected in the compressors, and a recording of the pressure pulsations in the presurge and surge regimes was made.

From the material considered we can realize the practical importance of the phenomena being studied, the need for further study of them, and the use of already available data in designing gas-turbine assemblies.

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